



## **Analysis of a cogeneration system based on solid oxide fuel cell and air source heat pump with novel heat recovery**

**Vialetto, G.; Noro, M.; Rokni, M.**

*Publication date:*  
2017

*Document Version*  
Peer reviewed version

[Link back to DTU Orbit](#)

### *Citation (APA):*

Vialetto, G., Noro, M., & Rokni, M. (2017). *Analysis of a cogeneration system based on solid oxide fuel cell and air source heat pump with novel heat recovery*. Paper presented at 12th Conference on Sustainable Development of Energy, Water and Environment Systems – SDEWES Conference, Dubrovnik, Croatia.

---

### **General rights**

Copyright and moral rights for the publications made accessible in the public portal are retained by the authors and/or other copyright owners and it is a condition of accessing publications that users recognise and abide by the legal requirements associated with these rights.

- Users may download and print one copy of any publication from the public portal for the purpose of private study or research.
- You may not further distribute the material or use it for any profit-making activity or commercial gain
- You may freely distribute the URL identifying the publication in the public portal

If you believe that this document breaches copyright please contact us providing details, and we will remove access to the work immediately and investigate your claim.

## **Analysis of a cogeneration system based on solid oxide fuel cell and air source heat pump with novel heat recovery**

G. Vialetto

Department of Management and Engineering, University of Padova, Vicenza, Italy  
e-mail: giulio@giuliovialetto.it

M. Noro

Department of Management and Engineering, University of Padova, Vicenza, Italy  
e-mail: marco.noro@unipd.it

M. Rokni

Technical University of Denmark, Copenhagen 2800, Denmark  
e-mail: mr@mek.dtu.dk

### **ABSTRACT**

Transition to energy systems with low greenhouse gases emissions is required by national and international organisations (e.g. EU Commission, Environmental Protection Agency and/or United Nations Framework Convention on Climate Change) with the aim of decreasing global warming. Different solutions have already been studied and proposed but more improvements could be achieved. In this paper a new heat recovery for micro-cogeneration system based on a Solid Oxide Fuel Cell and Air Source Heat Pump is presented with the main goal of improving efficiency on energy conversion for a residential building. The exhaust gases after the fuel cell are firstly used to heat water for heating/domestic water, and then they are mixed with the external air to feed the evaporator of the heat pump with the aim of increasing energy efficiency of the latter. This system configuration decreases the possibility of freezing of the evaporator as well, which is one of the drawbacks for air source heat pump in Nordic climates. Simulations are performed varying the external air temperature, air humidity and SOFC nominal power. COP increases up to more than 100 % when SOFC electric power is close to its nominal (50 kW), and/or inlet air has a high relative humidity (close to 100 %). Instead, the effect of mixing the exhausted gases with air may be negative (up to -25 %) when SOFC electric power is 20 kW and inlet air has 25 % relative humidity. Thermodynamic analysis is carried out to prove energy advantage of such a solution with respect to a traditional one, resulting to be between 37.5 % and 45 % in terms of primary energy.

### **KEYWORDS**

Adiabatic mixer, air source heat pump, energy performance, evaporator frosting, ground source heat pump, micro-cogeneration, Solid Oxide Fuel Cell

### **1. INTRODUCTION**

Solid Oxide Fuel Cells (SOFC) are a promising energy generation technology with high efficiency. Different configurations of SOFC based systems were proposed, for example combined with Rankine cycle [1], Organic Rankine Cycle [2], Stirling [3-5], gas turbine [6-7] and steam-injected gas turbine [8]. For example, in [9] a SOFC-gas turbine integrated system with the external steam reforming of ethanol to produce hydrogen for the SOFC was proposed. The effects of key operating parameters, such as pressure, fuel use and

turbomachinery efficiency, on the SOFC-gas turbine hybrid system performance were discussed. Instead, a new hybrid system that couples a solid state two-stage thermoelectric generator to a solid oxide fuel cell was proposed in [10]. The study optimized the number of thermoelectric elements among the top and bottom stages and a relationship between the operating current density of the SOFC and the dimensionless electric current of the optimized thermoelectric generator was derived.

SOFC have also high flexibility on the used fuel. Fuels such as ammonia, dimethyl ether (DME), methanol and ethanol were studied in [4], biomass gasification studied in [3] and [6], while municipal waste after gasification were simulated in [7]. In [11] two new trigeneration systems (producing power and heating alongside with cooling) based on solid oxide fuel cell fed by either the syngas or biogas were proposed. Through a parametric study, the effects of some key variables such as the current density and the fuel utilization factor in the systems' performance were investigated.

SOFC were already studied for micro-cogeneration systems, especially for the residential sector [12-14] in cases where the households require a system for both electricity and heat production with a power request wherein the heat to power ratio is changing during the time. In [15] one of the possible ways to integrate a heat pump with a micro-Combined Heat and Power based on Solid Oxide Fuel Cell technology was presented, in order to increase the overall efficiency of the building heating system. Three different geothermal heat pumping technologies (vapour compression, absorption and adsorption) were studied both from experimental and numerical model (by Trnsys) point of view. Simulations performed demonstrated that the overall yearly efficiency can be improved up to 30 %.

Micro-cogeneration was already proposed not only in household applications but also for electric mobility. The systems were based on different micro-cogeneration equipment, i.e. internal combustion engine coupled to natural gas boilers in [16-18], and SOFC in [19-22].

The present authors have already studied performance of heat pump systems applied to heating, ventilation and air conditioning plants, both with ground source heat pumps (GSHP) as in [23-24], and with air source heat pumps (ASHP) as in [25-26]. They proposed improvements of energy efficiency of SOFC systems coupled with GSHP for cold climates [27] and with ASHP for warmer climates [28]. In these cases, however, SOFC off-gases were used only to heat water for heating and/or domestic hot water production. An innovative operation strategy defined and analysed in [29] was used in those studies: it did not follow electrical or heating demand separately but considered both together. In that strategy, an electric equivalent load (EEL) parameter was defined, which summed the electrical demand for the user, the heat pump, and the electric heater (if used). EEL considered that the heating user demand was covered partly by the SOFC waste heat, partly by the heat pump, and partly by the integration heating system.

One of the main characteristic of SOFC systems is their high temperature in the exhausted gases, which can be used for different purposes. The off-gases temperature depends on the working condition of the system, and can be as high as e.g. 700 °C as shown in [4]. Heat recovery on the waste gases was studied both in a residential application [30] and in the industrial sector [31], but in a few cases, SOFC was combined with absorption chiller as proposed in [32].

Even after heat recovery, SOFC exhausted gases have an interesting enthalpy as they still have a temperature greater than 100 °C. In this paper authors propose a new heat recovery for micro-cogeneration SOFC system after the exhausted gases are firstly used to heat water for heating/domestic water: they are mixed with external air to feed the evaporator of an air source heat pump with the aim of increasing energy efficiency of the heat pump. Such a technique was already studied by the authors applied to a heating, ventilation and air conditioning plant set up by a gas engine driven heat pump in an existing historical building

[33]. This study presents the mathematical model of the system and the results of the first simulations. Here, an effective energy saving is proved while economic viability of such a system should be analysed in a future development of this work, based on an annual analysis on different climates. Anyway, economic analysis as given in the previous studies of the authors indicates potential positive results of such a system in both Northern [27] and Southern climate [28].

## 2. OVERVIEW OF THE SYSTEM

The system here proposed is set-up by a cogenerator (in this case a solid oxide fuel cell), an air source heat pump and an air mixer. Considering a typical residential building, SOFC provides for electricity while heat demand is satisfied by both the heat recovered from SOFC exhausted gases and the heat pump. In the configuration here proposed the exhausted gases further exchange heat by preheating inlet air at the evaporator section of ASHP. This may increase the overall performance of ASHP. It is worth to underline that in this study the possible direct use of exhausted gases heat is not considered as this possibility would have implied an air heating system for the served building. It is considered that the building is heated by a water based heating system only.

### 2.1 Solid Oxide Fuel Cell

SOFC is simulated using the simulation tool DNA (Dynamic Network Analyser) developed at DTU [4]. Natural gas as fuel for SOFC is here considered even if the effectiveness of other fuels (like ammonia, DME, ethanol or methanol) was proved in a previous study by the Authors [4]. The fuel cell cogenerator is set up by the following components (Figure 1):

- air compressors to compress the air necessary (106 kPa) for the fuel cell system;
- a catalytic partial oxidation (CPO) to convert the heavier hydrocarbons into  $\text{CH}_4$ ,  $\text{H}_2$  and  $\text{CO}$ ;
- a desulfurizer to remove the sulfur from the fuel and avoid fuel cell poisoning;
- heat exchangers to increase plant efficiency, preheating fuel and air using the off-fuel and off-air respectively (CP,RP,FP,AP), and to heat water for space heating as well as for domestic hot water (DHW) using the wasted off-gases (HEAT RECOVERY);
- a burner to increase enthalpy of the unused fuel out of the fuel cell;
- SOFC stacks, with performance calculated by the type developed at DTU Risø National Laboratory.

It is supposed that compressors' electricity consumption ( $\eta_{aux}$ ) is 1.5 % of the electricity produced, while inverter efficiency ( $\eta_{inv}$ ) is assumed to be 92 %. An overall transmitted efficiency ( $\eta_{trans}$ ) is defined considering both auxiliaries and inverter efficiencies (Equation 1):

$$\eta_{trans} = \eta_{aux}\eta_{inv} \quad (1)$$

Under such hypothesis the  $\eta_{trans}$  can be calculated to be 0.9068. Table 1 reports the thermodynamic benchmarks of a simulation of the SOFC system with 50 stack at full load (50 kW).

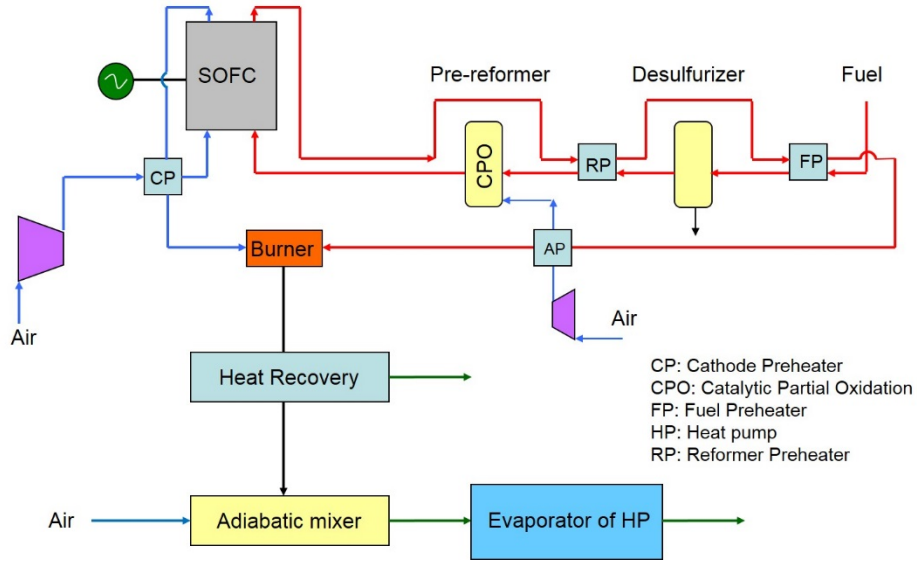


Figure 1. Representation of SOFC system ([27-28]). The air mixer to partly recover heat from the exhausted gases of the SOFC is connected after the Heat Recovery by point 1 (see next Figure 4).

Table 1. Efficiencies of the different components of the SOFC system.

Parameter	Value
SOFC (50 kW, full load), thermal efficiency	$\eta_{thermal,SOFC} = 0.4286$
SOFC (50 kW, full load), electrical efficiency	$\eta_{electrical,SOFC} = 0.5299$
SOFC (50 kW, full load), heat to power ratio	H/P = 0.8088
SOFC auxiliaries consumption, efficiency on electrical output	$\eta_{trans} = 0.9068$
Exhausted gas temperature	105 °C

A SOFC exhausted gas analysis was performed and Table 2 shows the results of such analysis. Note that sulphur formation in any form (such as  $SO_x$ ) is avoided thanks to the desulfuriser. If gas mixture is condensed, then condensate may not include any acid. At full-load it is calculated that the humidity ratio (water mass versus dry air mass) is  $0.342 \text{ kg}_{water}/\text{kg}_{dry\_air}$ .

Table 2. Mass composition of SOFC exhausted gas at full load, gas temperature 105 °C.

Composition	Gas percentage	Composition	Gas percentage
N <sub>2</sub>	56.76%	Ar	0.59%
Water	24.37%	NO	0.00%
CO <sub>2</sub>	12.90%	SO <sub>2</sub>	0.00%
O <sub>2</sub>	5.37%	NO <sub>2</sub>	0.00%

## 2.2 Heat pump

The heat pump plays an important role in the system. It recovers heat from exhausted gases at (relative) low temperature. In the previous studies, the authors proposed ground source heat pump for cold climates (such as in Denmark) [27] and air source heat pump for warmer climate (for example Mediterranean climates) [28] to be coupled to the fuel cell system to face the electricity and heating needs of a residential building. Usually GSHP are more expensive compared to ASHP due to drilling costs and associated heat exchanger. On the other hand, GSHPs have higher coefficient of performance (COP) because ground temperature is usually constant (or assumed) during the year. Air temperature, instead, varies between day and night times and month by month and therefore, ASHPs have variable COP during the year, and is typically lower during colder periods. It is advisable to consider that in climates where conditions with air temperature just above 0 °C (especially between 5 and 7 °C depending on the design of the finned coil) and relative humidity above 50 % are more frequent, possibility of freezing of the outdoor heat exchanger (evaporator) may lead to a decrease in seasonal performance of the heat pump (ice has poor heat transfer capability and reduces available area for air and so air mass flow rate). For such a reason defrost of evaporator section is periodically necessary. Defrosting can be performed by an auxiliary heat source (electrical resistance or gas burner) or reversing the cycle. In any case, defrosting is quite penalizing for the heat pump energy performance, as it increases its energy consumption. In this paper the authors propose to mix the exhausted gases exiting the SOFC (Figure 1 and point 1 in Figure 4) with outdoor air (point 2 in Figure 4) with the aim to enhance temperature of inlet air at the ASHP evaporator (point 3 in Figure 4) in order to prevent ice formation.

Heat pumps could be simulated using technical norms, for example UNI 11300-4 to consider different working temperature at condenser/evaporator and using EN 14825 for partial load of the heat pump in heating mode based on the studies of [34-36]. In this study, the authors propose a regression of technical datasheet (Figure 2, Figure 3, and Table 3).

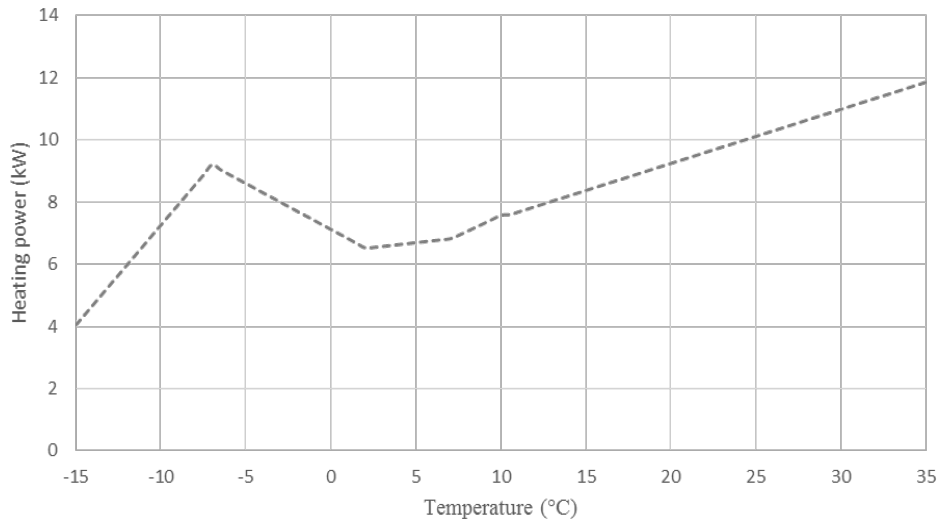


Figure 2. Technical datasheet, relation between nominal heating power and external air temperature ([37]).

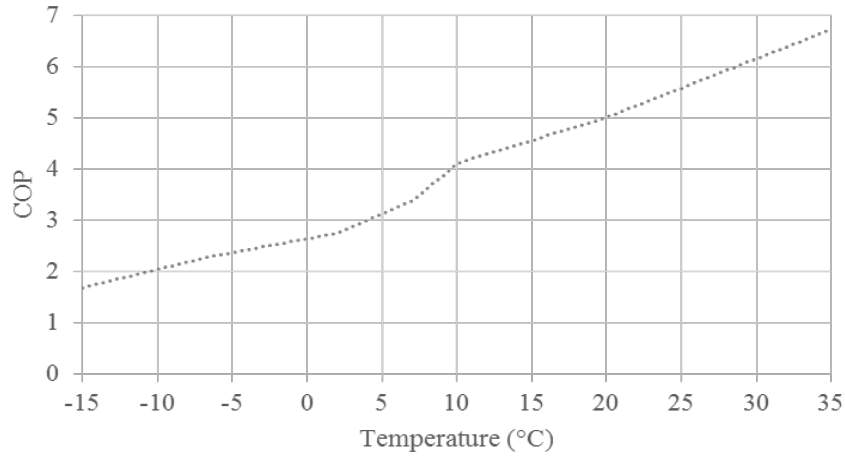


Figure 3. Technical datasheet, relation between nominal COP and external air temperature ([37]).

Table 3. Nominal conditions of ASHP.

Parameter	Value
ASHP Nominal Condition	External Air 2 °C - Condenser outlet Water 45 °C (A2W45)
P <sub>nom</sub> – Nominal heating power	7.7 kW
COP <sub>nom</sub> – Nominal COP	2.75

As previously mentioned, when external air relative humidity is higher than 50 % and temperature is just above 0 °C, freezing of evaporator may occur. Typically, freezing rate is maximum when air temperature is around 7 °C. A frost factor is considered [38] as multiplying penalty factor to decrease COP of the heat pump to take into account the periodic defrosting. Reference proposes a value that is defined as a function of outdoor air temperature (in the range of –10 to 10 °C) and relative humidity (in the range of 50-100 %): the penalty factor is lower given a lower outdoor air temperature (up to values just above 0 °C), and the higher relative humidity [38].

### 2.3 Adiabatic mixer and evaporator

The new issue proposed in this paper is mixing air with off-gases from the SOFC plant. The aim is to increase air temperature at evaporator inlet as much as possible to avoid freezing of evaporator and increasing the COP. Figure 4 represents the proposed air mixing system: SOFC exhausted gases (point 1) and outdoor air (point 2) are mixed in an adiabatic mixer. In some cases, humidity condensation may also occur (point 31). Mixture (point 3) is used in the evaporator delivering heat to the ASHP and then is discharged (point 4). Humidity condensation may occur also into the evaporator (point 41). It is supposed that no auxiliary flow inducing system such as a fan is necessary: the head pressure available from the fan of the heat pump model here referred (around 80 Pa) is supposed to be adequate to face the pressure drop of the mixer (and of the ducts and evaporator).

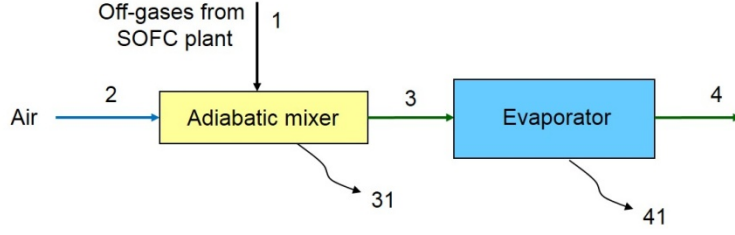


Figure 4. Air mixing system. Curves pointing down represent possible water condensation after the air heat exchange respectively in the mixer (point 31) and the evaporator (point 41).

A mathematical model is proposed to describe the components of the system according to equations regarding wet air proposed in [39]. Equations 2 describe the adiabatic mixer.

$$\begin{cases} m_{DA,1} + m_{DA,2} = m_{DA,3} \\ m_{DA,1} * W_1 + m_{DA,2} * W_2 = m_{DA,3} * W_3 + m_{l,31} \\ m_{DA,1} * h_1 + m_{DA,2} * h_2 = m_{DA,3} * h_3 + m_{l,31} * h_{l,31} \\ W_3 = \min\left(\frac{m_{DA,1} * W_1 + m_{DA,2} * W_2}{m_{DA,3}}, W_{sat,3}\right) \end{cases} \quad \text{Eq. (2)}$$

It is imposed:

- conservation of dry air mass ( $m_{DA,1}$ , point 1), dry exhausted gases ( $m_{DA,2}$ , point 2) and dry air mixture ( $m_{DA,3}$ , point 3);
- conservation of water mass flow rate.  $W_1, W_2, W_3$  are respectively the humidity ratio in points 1, 2 and 3,  $m_{l,3}$  is the liquid mass flow rate in case of humidity condensation in the adiabatic mixer;
- conservation of energy.  $h_1, h_2, h_3$  are respectively the specific enthalpy in 1, 2, 3, the values depend on air temperature and humidity ratio.  $h_{l,3}$  is the enthalpy of condensate water.

$W_{sat,3}$  is the humidity ratio in state 3 in saturation condition, depending only on air temperature in 3 (total pressure is assumed to be 101325 Pa). If  $W_{sat,3}$  is lower than weighted average of humidity ratio in states 1 and 2 condensation occurs.

Equations 3 describe the evaporator.

$$\begin{cases} m_{DA,3} = m_{DA,4} \\ m_{DA,3} * W_3 = m_{DA,4} * W_4 + m_{l,41} \\ m_{DA,3} * h_3 = m_{DA,4} * h_4 + m_{l,41} * h_{l,41} + Q \\ W_4 = \min(W_3, W_{sat,4}) \end{cases} \quad \text{Eq. (3)}$$

It is imposed:

- conservation of dry air mixture mass.  $m_{DA,3}, m_{DA,4}$  are respectively the dry mass flow rate in states 3 and 4;
- conservation of water mass.  $W_3, W_4$  are the humidity ratio values respectively in 3 and 4,  $m_{l,4}$  is the liquid mass flow rate in case of humidity condensation in the evaporator;
- conservation of energy.  $h_3, h_4$  are respectively the specific enthalpy in 3 and 4, they depend on air temperature and humidity ratio.  $h_{l,4}$  is the enthalpy of condensate water and  $Q$  is the heat absorbed by the refrigerant at the heat pump evaporator.

$W_{sat,4}$  is the humidity ratio in 4 in saturation condition, depending only on air temperature in 4. If  $W_{sat,4}$  is lower than humidity saturation in state 3 condensation occurs.



The heat pump is expected to increase its performances because of the higher air enthalpy at the evaporator inlet. This is due to both the higher temperature and the higher humidity ratio (both sensible and latent terms contribute to the enhancement of enthalpy due to the adiabatic mixer).

### 3. SIMULATION OF THE SYSTEM

A steady-state analysis is performed with the aim of studying the energy performance of the system at different operation conditions. The analysis is performed by varying the dry bulb air temperature from  $-7.5\text{ }^{\circ}\text{C}$  to  $15\text{ }^{\circ}\text{C}$  with a step of  $2.5\text{ }^{\circ}\text{C}$ , relative humidity from 25 % to 100 % with a step of 25 %, and SOFC nominal electric power from 20 to 50 kW (step of 10 kW). For the sake of brevity, only two couples of values, namely outdoor air relative humidity and SOFC nominal electric power, are presented in next section 4 to compare very different situations: 25 % - 20 kW, and 100 % - 50 kW. Varying SOFC nominal electrical power is advisable because it affects the exhausted gases flow rate entering the adiabatic mixer. A 6 kW nominal heating power ASHP is chosen in order to have consistent mass flow rates between heat pump evaporator and SOFC.

The simulations allow to calculate air temperature at the evaporator outlet (point 4 in Figure 4), as well as the COP of the heat pump. The main aim of the proposed system is to increase the COP of the ASHP. As previously mentioned, COP is a function of air temperature and relative humidity. A high value of the latter is useful to improve the COP because of the condensation of latent heat that increases the heat exchange inside the evaporator, under assumption that the finned coil surface temperature is not below  $0\text{ }^{\circ}\text{C}$ . Unlike, frost may grow in the fins reducing heat exchange between air and refrigerant. As already cited in section 2.2, this may occur more frequently when air temperature in the evaporator inlet is between  $5\text{ }^{\circ}\text{C}$  and  $9\text{ }^{\circ}\text{C}$ . In such case, a defrost factor has to be considered [40] to take into account the penalization of COP due to periodic defrosting of the evaporator with finned coil (e.g. by reversing the cycle). Analysis of COP variation between a traditional ASHP (without the adiabatic mixer) and the current system is performed and presented for four very different representative cases by a combination of external air relative humidity and SOFC nominal power (25 % - 20 kW, 25 % - 50 kW, 100 % - 20 kW, 100 % - 50 kW) (Equation 4):

$$COP_{variation} = \left( \frac{COP_{innov,sys}}{COP_{trad,sys}} - 1 \right) \cdot 100 \quad \text{Eq. (4)}$$

$COP_{variation}$  higher than 0 (zero) means that the current system has a higher COP than the traditional one, and therefore ASHP performs better.

Analysis on primary energy saving (%PES) is also proposed, wherein the innovative system is compared with a traditional one with separate production of heat (boiler) and electricity (national grid) in terms of primary energy (PE). Considering that the system here proposed has a net available electricity generation  $E_{ava}$  (that is the difference between SOFC net electric power and ASHP consumption) and a heat generation  $H_{ava}$  (that is the sum of heat cogenerated by SOFC and generated by ASHP), %PES benchmark is defined as (Equation 5):

$$\%PES = \left( 1 - \frac{PE_{innov,sys}}{PE_{trad,sys}} \right) \cdot 100 = \left( 1 - \frac{\frac{E_{ava}}{\eta_{ele}} + \frac{H_{ava}}{\eta_{boiler}}}{F_{SOFC}} \right) \cdot 100 \quad \text{Eq. (5)}$$

where  $F_{SOFC}$  is the fuel (primary energy) consumption of SOFC,  $\eta_{ele}$  is the global electric efficiency from grid (assumed to be 0.435), and  $\eta_{boiler}$  is the efficiency of boiler for heat production (assumed to be 0.9). Such a definition is consistent with that of the primary energy

saving of cogeneration systems as referenced in the 2012/27/EU Energy Efficiency Directive [41] and Directive 2004/8/EC on promotion of cogeneration [42].

## 4. RESULTS AND DISCUSSION

Main results for the representative cases described in previous section 3 are here reported in terms of evaporator outlet temperature, Coefficient Of Performance and Primary Energy Saving of the proposed system.

### 4.1 Evaporator outlet air temperature

Firstly, the difference in evaporator outlet temperature (airside) between an ASHP standalone (only ASHP) and ASHP-SOFC integrated system is outlined (Figure 5). In both cases considered as previously described (case 1: 25 % - 20 kW, and case 2: 100 % - 50 kW), temperature at the evaporator outlet is higher in the ASHP-SOFC system because of the positive effect of high temperature of the exhausted gases from SOFC. The higher the SOFC electric power, the higher the temperature difference between the two systems is found to be. The reason is due to the increasing of airflow rate. For example, at 0 °C external air temperature  $T_4$  is 3 °C higher with ASHP-SOFC system with respect to ASHP only in case 1, whereas at same conditions  $T_4$  is 10 °C higher in case 2.

Even if the increasing of temperature of discharged gases (point 4, Figure 4) is proved, COP may increase or not. If evaporator temperature is between -7.5 °C and 10 °C and relative humidity is higher than 50 %, then the frost factor shall be considered, which can affect the energy performance of the heat pump.

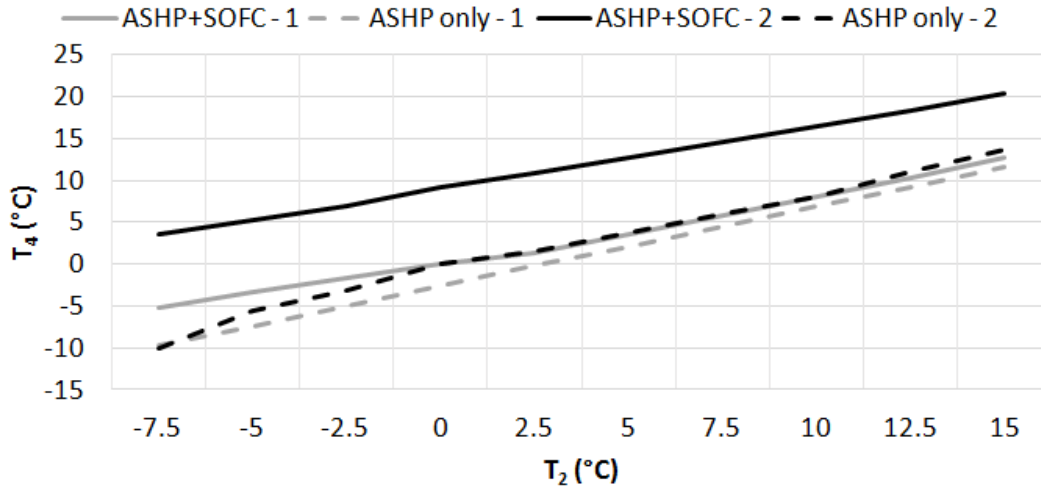


Figure 5. Evaporator outlet air temperature ( $T_4$ ) in function of external air temperature ( $T_2$ ) in the two cases (air relative humidity - SOFC nominal electric power). Case 1: 25 % - 20 kW; Case 2: 100 % - 50 kW.

### 4.2 Coefficient of Performance

Figure 6 compares the COP of the presented system with that of an ASHP only, considering the two very different cases previously described: low external air relative humidity with low SOFC nominal electric power (respectively 25 % and 25 kW), and high external air relative humidity with high SOFC nominal electric power (respectively 100 % and 50 kW). It is apparent that the system proposed here is not always advantageous. The latent heat contribution of SOFC exhausted gases may be greater than the sensible one and therefore more frequent defrosting is requested when air temperature is in the critical range (5–9 °C as

already stated). The higher weight of the frost factor may decrease the COP of the innovative system. For low humidity and low power (case 1), the proposed system has lower COP than traditional one when external air temperature is lower than about 8.5 °C, ranging from 2 to 3 for ASHP only and from 1.7 to 3 for ASHP-SOFC. However, in the case of 100 % - 50 kW, the higher exhausted mass flow rate (due to the higher electrical power) allows COP of the proposed system to be always higher (between 2.1 and 4.8) than COP of the traditional system (between 1.7 and 4).

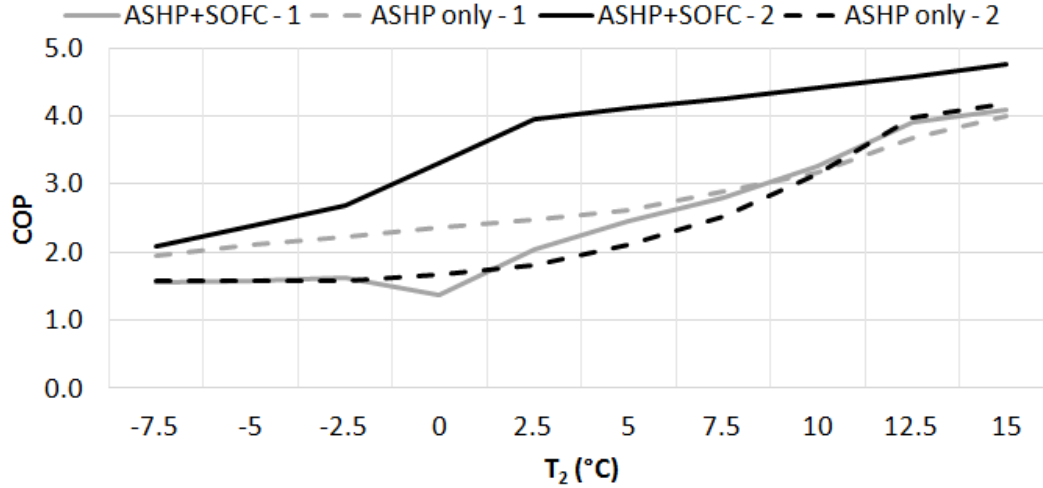


Figure 6. COP in function of external air temperature ( $T_2$ ) in the two cases (air relative humidity - SOFC nominal electric power). Case 1: 25 % - 20 kW; case 2: 100 % - 50 kW.

Figure 7 depicts  $COP_{variation}$  with varying outdoor air temperature for four very different representative cases by a combination of external air relative humidity and SOFC nominal power.  $COP_{variation}$  is influenced by:

- inlet air relative humidity. Given a fixed amount of SOFC power,  $COP_{variation}$  increases with increasing relative humidity of the air. According to Figure 7, the present system with 20 kW SOFC nominal power has a COP lower than the traditional one ( $COP_{variation}$  is lower than 0) at 0 °C and 25 % external air condition.  $COP_{variation}$  becomes positive if inlet air has higher relative humidity (e.g. 100 %);
- SOFC electric power. The higher the SOFC power, the higher the mass flow rate of exhausted gases, so the higher the temperature of the gases at the outlet of the adiabatic mixer. This parameter has a strong effect on system performances. Figure 7 shows that for inlet air temperature equal to 2.5 °C  $COP_{variation}$  increases from 30 % up to 110 % when considering a SOFC power of 20 kW and 50 kW, respectively.

The main conclusion of this reasoning is that the adiabatic mixer has a positive effect on the heat pump COP when outdoor air has a high relative humidity and when SOFC electric power is high. When SOFC power is 50 kW, COP of heat pump is always improved by the mixer. If a 20 kW SOFC is used, then the present system has a higher COP only when relative humidity of the inlet air is close to 100 %.

Figure 7 shows another interesting aspect of the system in which the higher air relative humidity and SOFC power, the lower air temperature at which maximum  $COP_{variation}$  occurs. If inlet air temperature is above 12.5 °C, the adiabatic mixer is not useful at all.

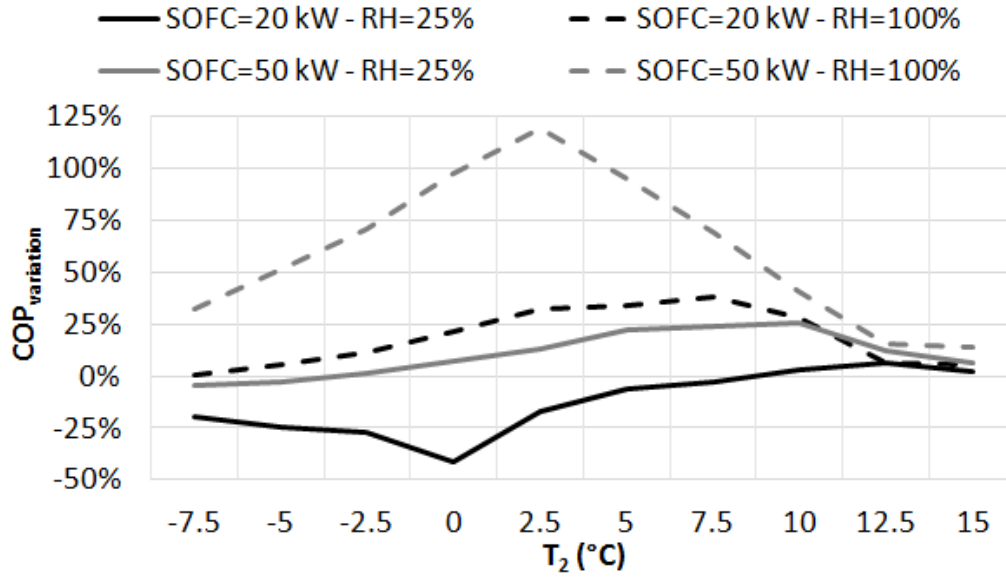


Figure 7. COP variation varying the external inlet air temperature for four very different cases in terms of SOFC nominal power and air relative humidity.

#### 4.3 Primary Energy Saving

As previously mentioned, %*PES* is defined to quantify energy saving from the proposed system and compared to the traditional solution (system). Figure 8 depicts %*PES* as a function of the outdoor air temperature for the same four different cases as in Figure 7 (very different cases but representative). The proposed system allows a primary energy saving in the range of 37.5 % – 45 %. The system with relatively small SOFC power presents lower %*PES* compared to the case with relatively high SOFC power, only for temperature below 2.5 °C and when relative humidity is low (25 %). Such critical value of air temperature decreases to 0.5 °C when humidity is high (close to 100 %).

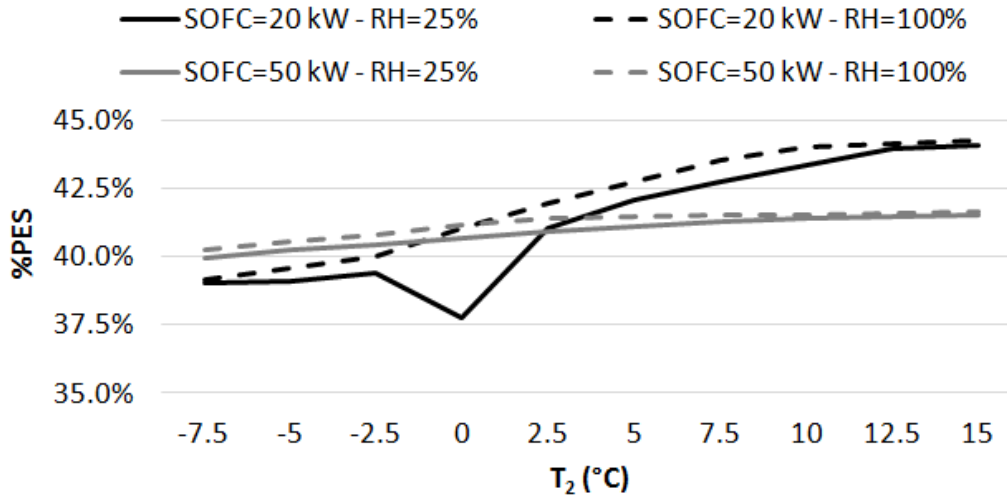


Figure 8. Primary energy saving varying the external inlet air temperature for four very different cases in terms of SOFC nominal power and air relative humidity.

Note that the higher the relative humidity, the higher %*PES* in the given temperature range of  $T_2$  (dotted lines are always above continuous lines in Figure 8). It is also worth to note that

primary energy saving depends also on the partial load operation of SOFC. This is due to variation of power ratio, and thereby the efficiency of SOFC.

## 5. CONCLUSIONS

In this study, a cogeneration system based on SOFC fed by natural gas (producing both electricity and heat) and ASHP with an advanced heat recovery system (to increase overall heat production) is analyzed. Outdoor air entering the evaporator of heat pump is mixed with exhausted gases from the SOFC plant with the aim of increasing evaporator temperature and thereby reducing possibility of freezing. Such conditions allow increasing the coefficient of performance for the heat pump.

Simulations are performed varying the external air temperature, air humidity and SOFC nominal power. Thermodynamic analysis shows that in some cases the effect of mixing the exhausted gases with air is negative: when SOFC electric power is lower in comparison to its nominal power (50 kW) and/or inlet air has a low relative humidity, COP decreases up to 35 %. Instead, COP increases up to 100 % when SOFC electric power is close to its nominal, and/or inlet air has a high relative humidity. A comparison based on primary energy consumption between the system proposed here and a traditional one with separate production (electricity demand is covered by the national grid and heat demand is produced separately by a boiler) proves that significant savings can be achieved (between 37.5 % and 45 %).

Results of the present research work show that ASHP performance could be increased considerably during cold season for climates with high relative humidity. Such results quantify the coefficient of performance and primary energy saving of the SOFC-ASHP integrated system, giving the base for a further study of an energy scenario where distributed energy generation is proposed, especially for household application. Similar studies can be carried out considering air temperature and air humidity distributions for different climates in order to quantify energy saving for different areas.

## REFERENCES

- [1] Rokni, M., Thermodynamic analysis of an integrated solid oxide fuel cell cycle with a rankine cycle, *Energy Conversion and Management*, Vol. 51, pp 3724-3732, 2010.
- [2] Pierobon, L., Rokni, M., Larsen, U., Haglind, F., Thermodynamic analysis of an integrated gasification solid oxide fuel cell plant combined with an organic Rankine cycle, *Renewable Energy*, Vol. 60, pp 226-234, 2012.
- [3] Rokni, M., Thermodynamic and thermoeconomic analysis of a system with biomass gasification, solid oxide fuel cell (SOFC) and Stirling engine, *Energy*, Vol. 76, pp 19-31, 2014.
- [4] Rokni, M., Thermodynamic analysis of SOFC (solid oxide fuel cell) – Stirling hybrid plants using alternative fuels, *Energy*, Vol. 61, pp 87–97, 2013.
- [5] Hosseinpour, J., Sadeghi, M., Chitsaz, A., Ranjbar, F., Rosen, M.A., Exergy assessment and optimization of a cogeneration system based on a solid oxide fuel cell integrated with a Stirling engine, *Energy Conversion and Management*, Vol. 143, pp 448-458, 2017.
- [6] Bang-Møller, C., Rokni, M., Thermodynamic performance study of biomass gasification, solid oxide fuel cell and micro gas turbine hybrid systems, *Energy Conversion and Management*, Vol. 51, pp 2330-2339, 2010.
- [7] Bellomare, F., Rokni, M., Integration of a municipal solid waste gasification plant with solid oxide fuel cell and gas turbine, *Renewable Energy*, Vol. 55, pp 490-500, 2013.
- [8] Mazzucco, A., Rokni, M., Thermo-economic analysis of a solid oxide fuel cell and steam injected gas turbine plant integrated with woodchips gasification, *Energy*, Vol. 76, pp 114-129, 2014.

- [9] Saebea, D., Magistri, L., Massardo, A., Arpornwichanop, A., Cycle analysis of solid oxide fuel cell-gas turbine hybrid systems integrated ethanol steam reformer: Energy management, *Energy*, Vol. 127, pp 743-755, 2017.
- [10] Zhang, H., Xu, H., Chen, B., Dong, F., Ni, M., Two-stage thermoelectric generators for waste heat recovery from solid oxide fuel cells, *Energy*, <https://doi.org/10.1016/j.energy.2017.05.005>
- [11] Mortazaei, M., Rahimi, M., A comparison between two methods of generating power, heat and refrigeration via biomass based Solid Oxide Fuel Cell: A thermodynamic and environmental analysis, *Energy Conversion and Management*, Vol. 126, pp 132-141, 2016.
- [12] Liso, V., Zhao, Y., Brandon, N., Nielsen, M. P., Kær, S. K., Analysis of the impact of heat-to-power ratio for a SOFC-based mCHP system for residential application under different climate regions in Europe, *International Journal of Hydrogen Energy*, Vol. 36, pp 13715-13726, 2011.
- [13] Bompard, E., Napoli, R., Wan, B., Orsello, G., Economics evaluation of a 5 kW SOFC power system for residential use, *International Journal of Hydrogen Energy*, Vol. 33, pp 3243-3247, 2008.
- [14] Hawkes, A., Leach, M., Solid oxide fuel cell systems for residential micro-combined heat and power in the UK: Key economic drivers, *Journal of Power Source*, Vol. 149, pp 72-83, 2005.
- [15] Frazzica, A., Briguglio, N., Sapienza, A., Freni, A., Brunaccini, G., Antonucci, V., Ferraro, M., Analysis of different heat pumping technologies integrating small scale solid oxide fuel cell system for more efficient building heating systems, *International Journal of Hydrogen Energy*, Vol. 40(42), pp 14746-14756, 2015.
- [16] Angrisani, G., Canelli, M., Roselli, C., Sasso, M., Integration between electric vehicle charging and micro-cogeneration system, *Energy Conversion and Management*, Vol. 98, pp. 115-126, 2015.
- [17] Rosato, A., Sibilio, S., Scorpio, M., Dynamic performance assessment of a residential building-integrated cogeneration system under different boundary conditions. Part I: Energy analysis, *Energy Conversion and Management*, Vol. 79, pp 731-748, 2014.
- [18] Ribberink, H., Entchev, E., Exploring the potential synergy between micro-cogeneration and electric vehicle charging, *Applied Thermal Engineering*, Vol. 71, pp 677-685, 2014.
- [19] Wakui, T., Wada, N., Yokoyama, R., Feasibility study on combined use of residential SOFC cogeneration system and plug-in hybrid electric vehicle from energy-saving viewpoint, *Energy Conversion and Management*, Vol. 60, pp 170-179, 2012.
- [20] Tanaka, T., Kamiko, H., Bando, T., Zaffirah, A., Kakimoto, N., Inui, Y., Maeda, T., Energetic analysis of SOFC co-generation system integrated with EV charging station installed in multifamily apartment, *International Journal of Hydrogen Energy*, Vol. 39, pp 5097-5104, 2014.
- [21] Vialetto, G., Noro, M., Rokni, M., Combined micro-cogeneration and electric vehicle system for household application: An energy and economic analysis in a Northern European climate, *International Journal of Hydrogen Energy*, Vol. 42(15), pp 10285-10297, 2017.
- [22] Vialetto, G., Noro, M., Rokni, M., Thermodynamic investigation of a shared cogeneration system with electrical cars for northern Europe climate, *Journal of*

*Sustainable Development of Energy, Water and Environment Systems*,  
<http://dx.doi.org/10.13044/j.sdewes.d5.0162>.

- [23] Busato, F., Lazzarin, R., Noro, M., Two years of recorded data for a multisource heat pump system: A performance analysis, *Applied Thermal Engineering*, Vol. 57, pp 39-47, 2013.
- [24] Busato, F., Lazzarin, R., Noro, M., Ground or solar source heat pump systems for space heating: Which is better? Energetic assessment based on a case history, *Energy and Buildings*, Vol. 102, pp 347-356, 2015.
- [25] Busato, F., Lazzarin, R., Noro, M., Ten years history of a real gas driven heat pump plant: energetic, economic and maintenance issues, *Applied Thermal Engineering*, Vol. 31, pp 1648-1654, 2011.
- [26] Busato, F., Lazzarin, R., Noro, M., Energy and economic analysis of different heat pump systems for space heating, *International Journal of Low Carbon Technologies*, Vol. 7(2), pp 104-112, 2012.
- [27] Vialetto, G., Rokni, M., Innovative household systems based on solid oxide fuel cells for a northern European climate, *Renewable Energy*, Vol. 78, pp 146-156, 2015.
- [28] Vialetto, G., Noro, M., Rokni, M., Innovative household systems based on solid oxide fuel cells for the Mediterranean climate, *International Journal of Hydrogen Energy*, Vol. 40, No. 41, pp 14378-14391, 2015.
- [29] Kavvadias, K.C., Tosios, A.P., Maroulis, Z.B., Design of a combined heating, cooling and power system: sizing, operation strategy selection and parametric analysis, *Energy Conversion and Management*, Vol. 51(4), pp 833-845, 2010.
- [30] Wang, K., Li, N., Peng, J., Wang, X., Wang, C., Wang, M., A highly efficient solution for thermal compensation of ground-coupled heat pump systems and waste heat recovery of kitchen exhaust air, *Energy and Buildings*, Vol. 138, pp 499-513, 2017.
- [31] Oluleye, G., Smith, R., Jobson, M., Modelling and screening heat pump options for the exploration of low grade waste heat in process sites, *Applied Energy*, Vol. 169, pp 267-286, 2016.
- [32] Zink, F., Lu, Y., Schaefer, L., A solid oxide fuel cell system for buildings, *Energy Conversion & Management*, Vol. 48, pp 809-818, 2017.
- [33] Lazzarin, R., Noro, M., District heating and gas engine heat pump: economic analysis based on a case study, *Applied Thermal Engineering*, Vol. 26, pp. 193-199, 2006.
- [34] UNI/TS 11300-4:2010 “Renewable energy and other generation systems for space heating and domestic hot water production” (Italian), Ente Nazionale Italiano di Unificazione, 2010.
- [35] EN 14825:2008. Air conditioners, liquid chilling packages and heat pumps, with electrically driven compressors, for space heating and cooling - Testing and rating at part load conditions and calculation of seasonal performance. European Committee for Standardisation, 2008.
- [36] Busato, F., Lazzarin, R., Noro, M., Energy and economic analysis of different heat pump systems for space heating, *International Journal of Low-Carbon Technologies*, Vol. 7, pp 104-112, 2012.
- [37] Technical datasheet from Viessmann – Vitocal 200-A. Available at <http://downloads.viessmannitalia.it/documents/5820-437-052015.pdf> - December 2016
- [38] Busato, F., Lazzarin, R., Minchio, F., Noro, M., *Sorgenti termiche delle pompe di calore. Aspetti tecnici, economici e normativi* (Heat sources of heat pumps. Technical, economic and standard aspects, in Italian), Editoriale Delfino, Milano, ISBN 978-88-97323-16-7, 2012.
- [39] ASHRAE, *Fundamentals, ASHRAE Handbook– Chapter 1 – Psychometrics*, 2009.

- [40] Lazzarin, R., *Pompe di calore. Parte teorica, parte applicativa* (Heat Pumps. Theoretic Part, Application Part, in Italian), SGE, Padova, 2011.
- [41] Directive 2012/27/EU of the European Parliament and of the Council of 25 October 2012 on energy efficiency
- [42] Directive 2004/8/EC of the European Parliament and of the Council of 11 February 2004 on promotion of cogeneration based on a useful heat demand in the internal energy market